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F2D
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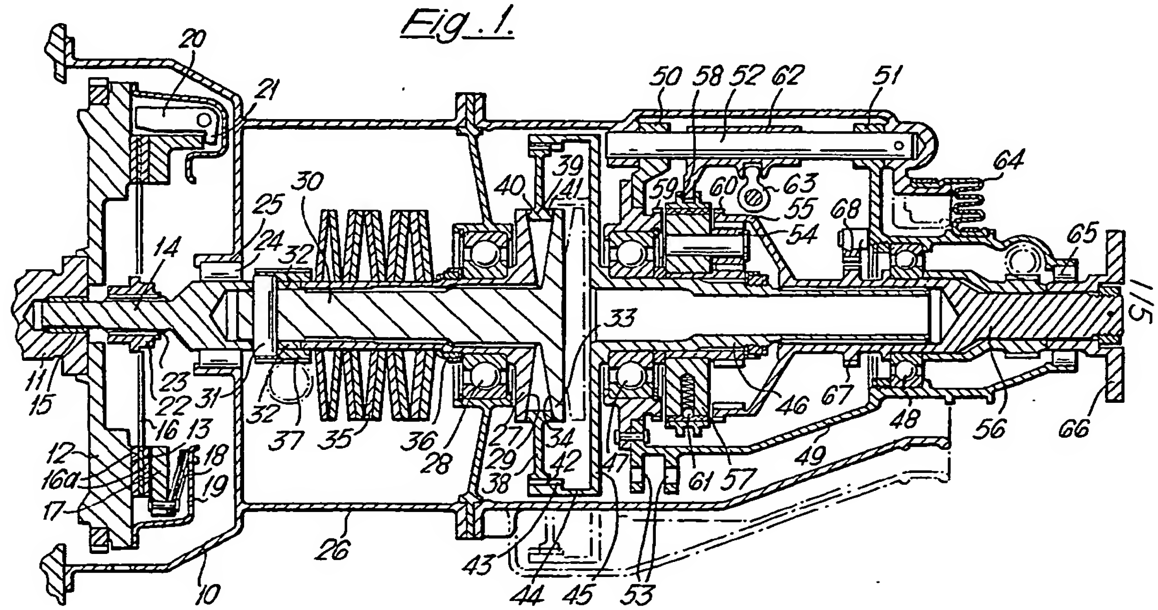
(54) Continuously variable transmission

(57) An automotive transmission comprises a clutch, a continuously variable frictional transmission and an epicyclic transmission connected in series. The frictional transmission includes a driving disc 27 having a frustoconical drive surface and a driving annulus 39 having a corresponding frustoconical drive surface which frictionally engages with the surface of the drive disc. The disc and annulus are relatively movable between a position in which they are coaxial and afford a unity drive ratio, and a position in which their axes are displaced and a different drive ratio is provided. The frictional transmission in turn drives the sun wheel of an epicyclic transmission having planet wheels 54 in a carrier 57 and an annulus 55 driving the output. Engagement means for the planet carrier provides direct drive, geared-down reverse, and neutral. The frictional transmission is adjusted automatically in response to engine speed and accelerator position. Also described is an arrangement in which an exhaust-driven turbine augments the power output of a piston engine through a frictional transmission of the kind described.

Fig. 1.



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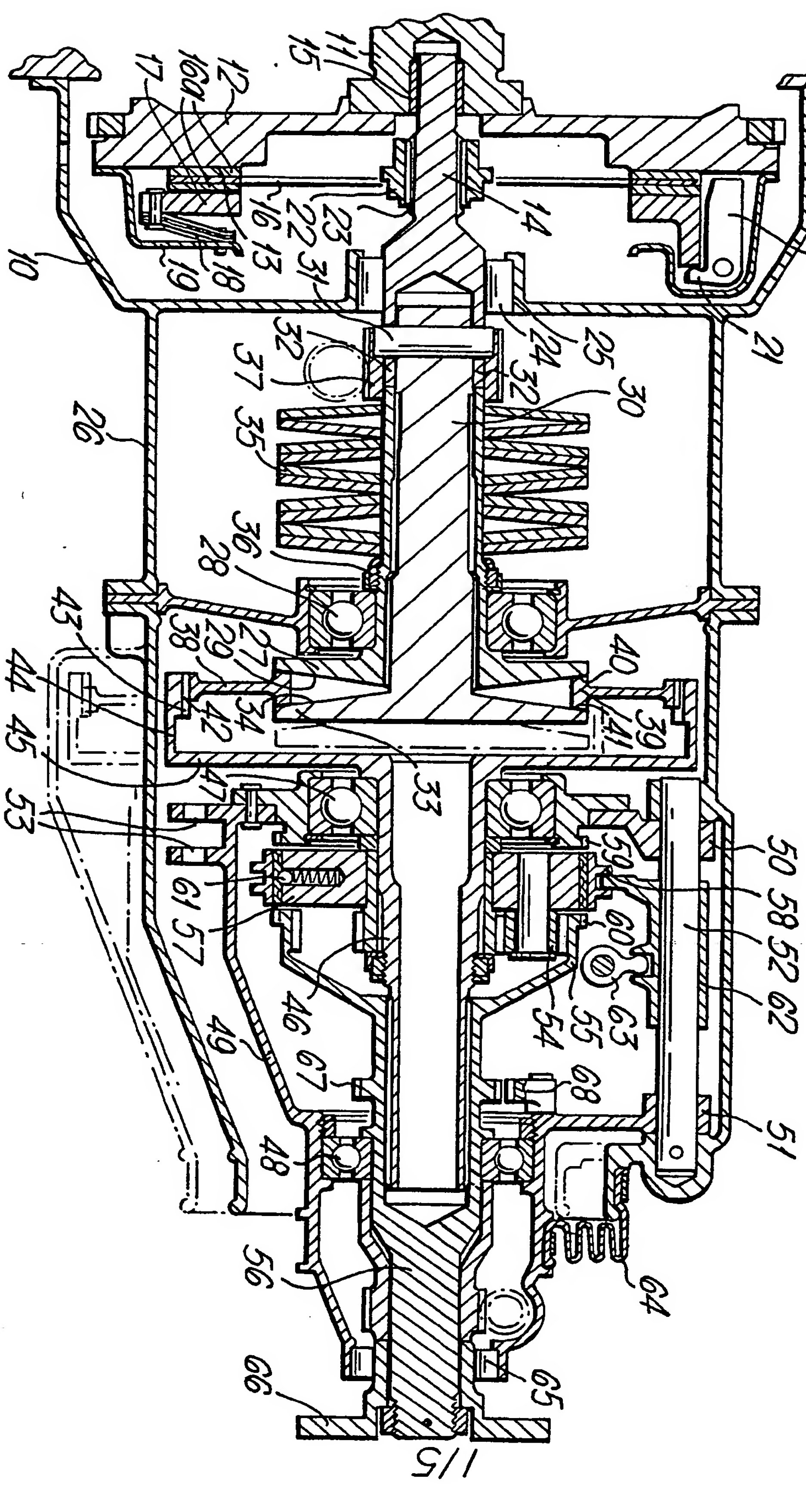

Fig. 1.

Fig. 2.

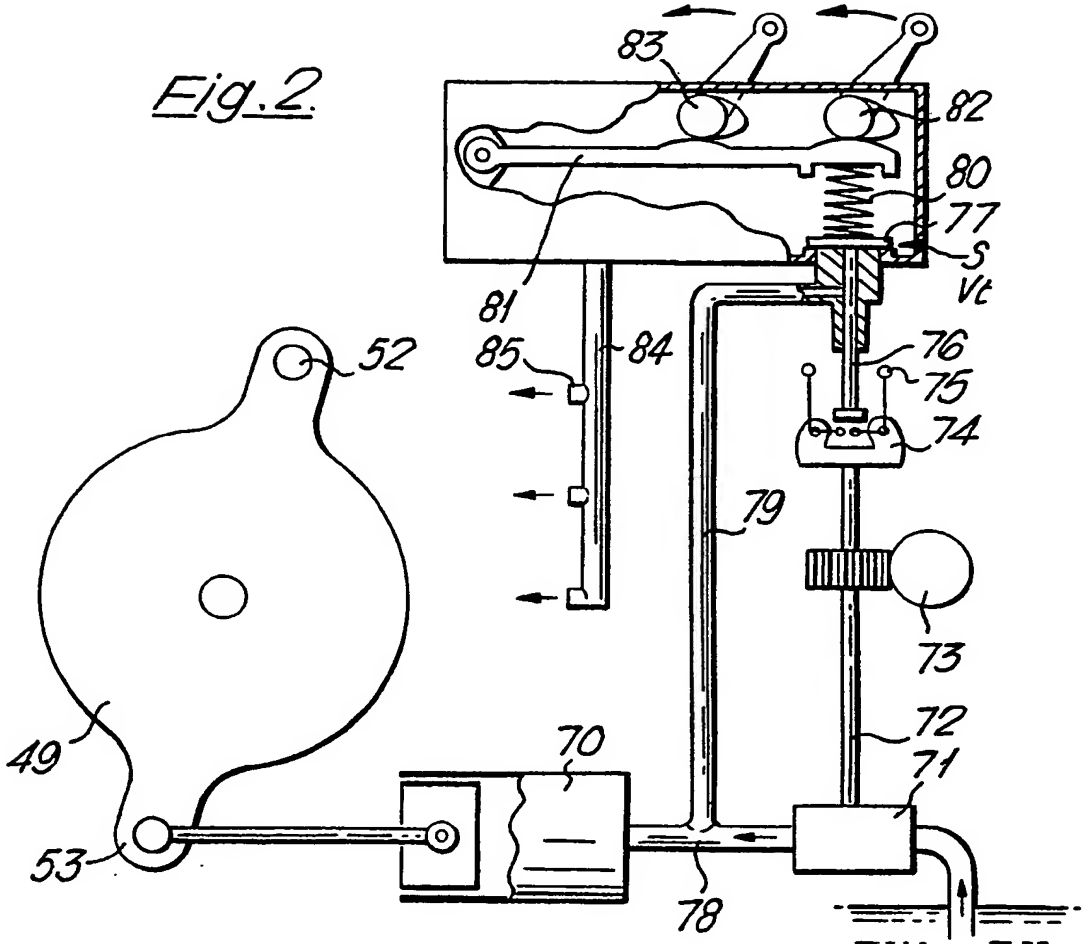
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Fig. 3.

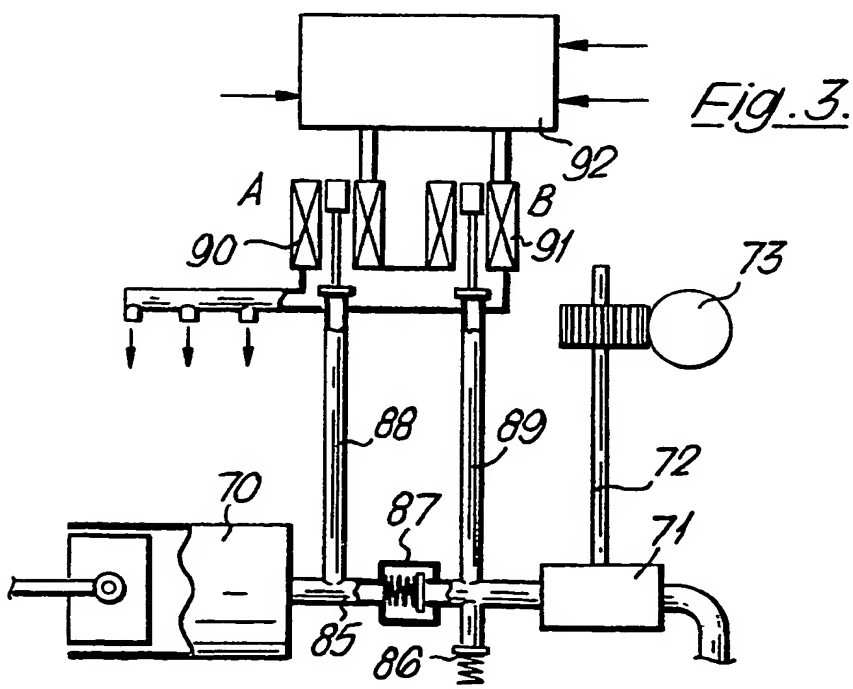
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Fig.4.

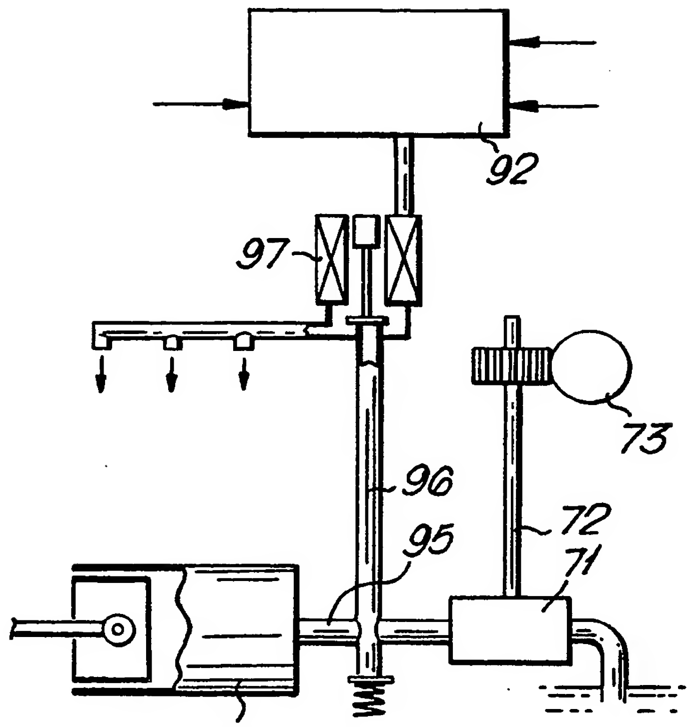
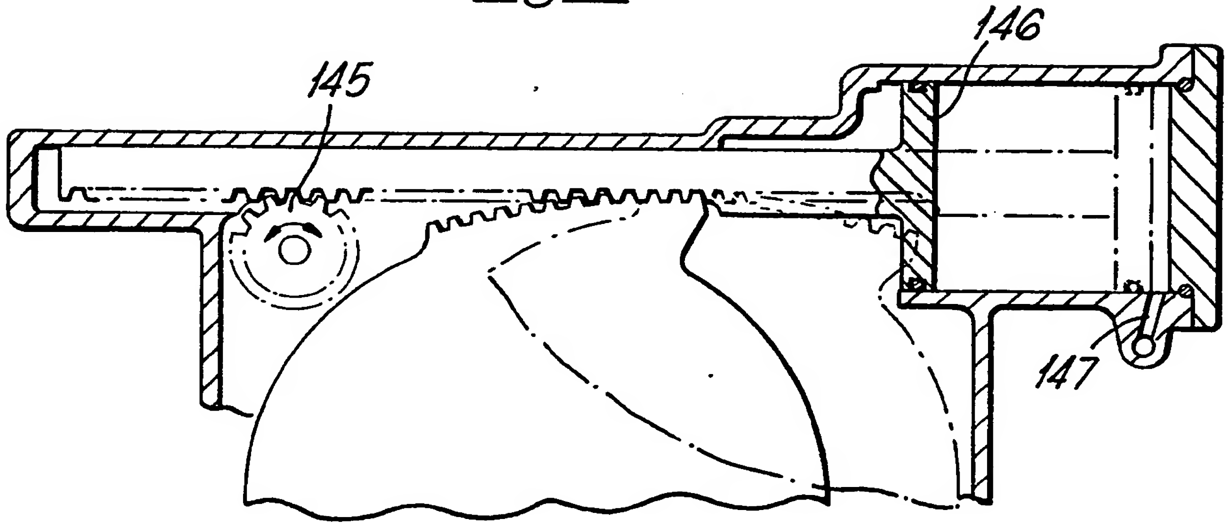


Fig.6.



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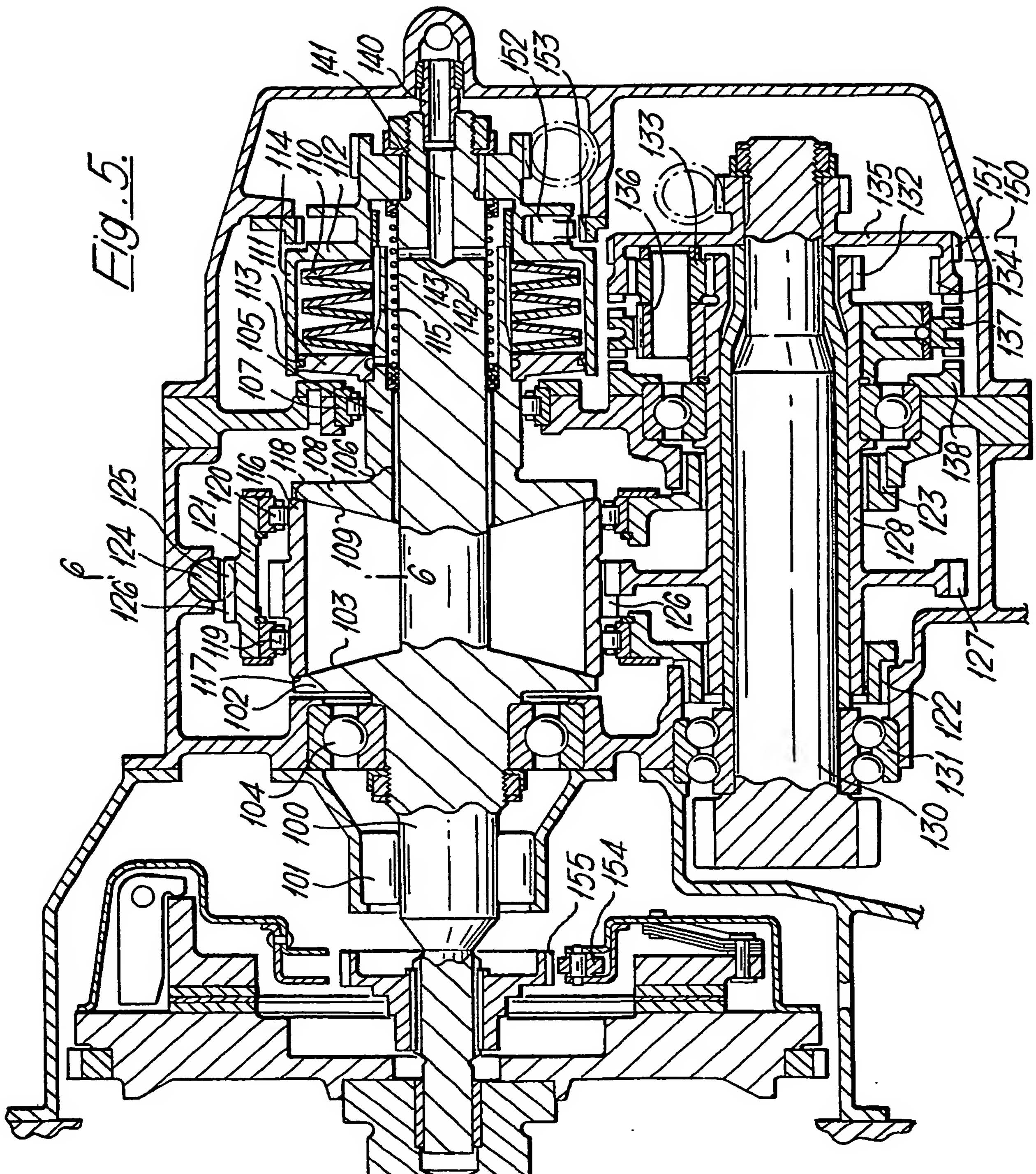

Fig. 5.

Fig. 7.

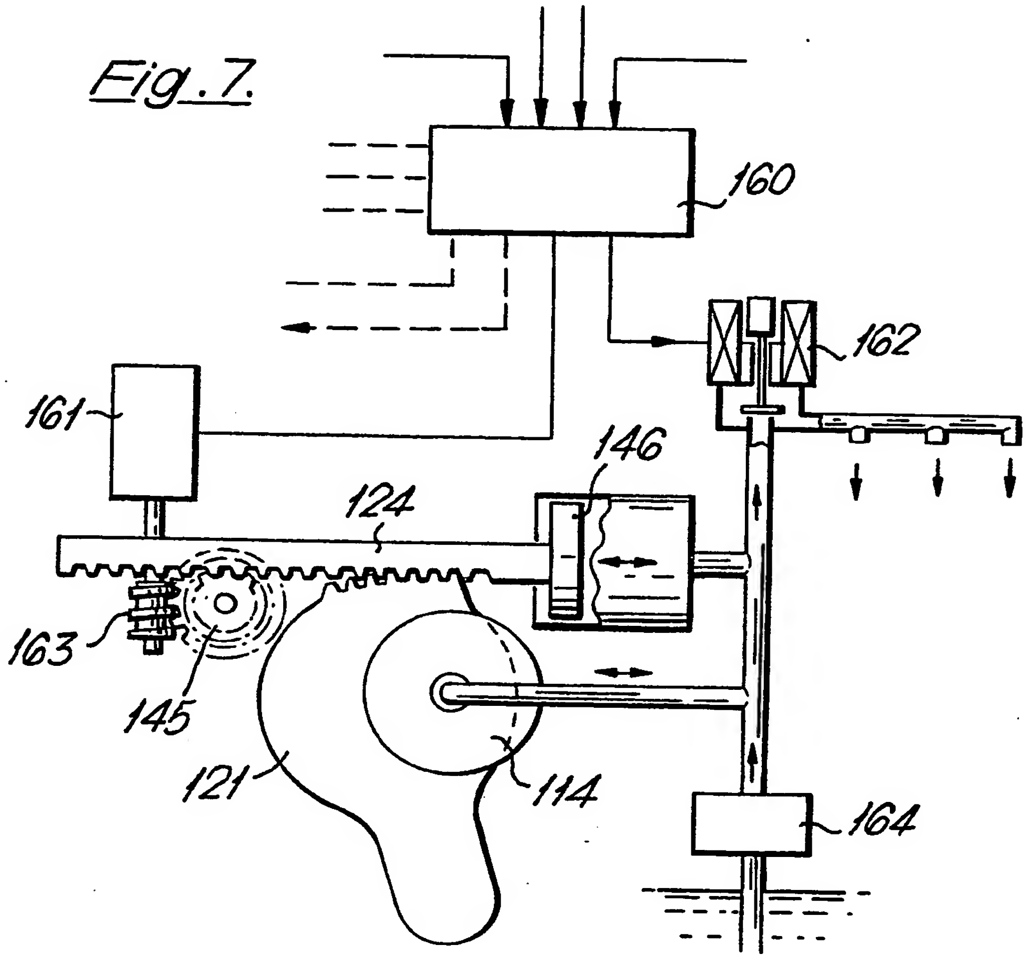
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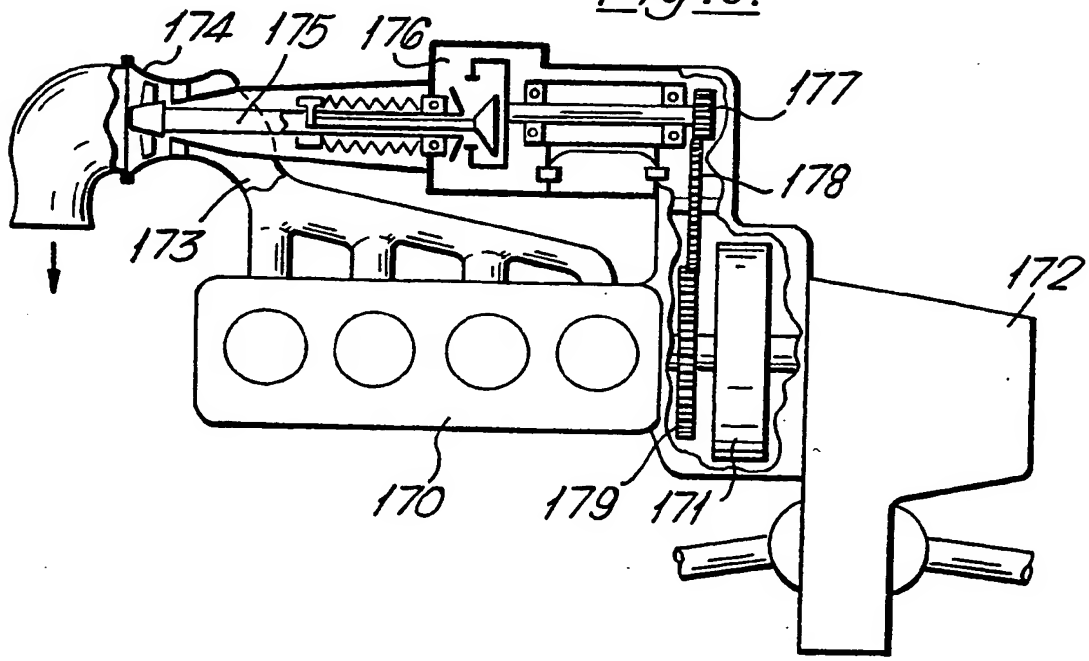
Fig. 8.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SPECIFICATION

Automotive drive transmission

This invention relates to an automotive drive transmission.

It is presently the practice to use mechanical automatic drive transmissions for motor cars and the like which normally include a torque converter and a series of change speed gears. There are various difficulties arising from these conventional constructions. Thus it is very difficult to avoid the changing of the gears producing a jolt to the vehicle and the torque converter is a relatively inefficient device.

The present invention relates to an automotive automatic transmission in which a stepless change of ratio is provided.

According to the present invention an automotive transmission comprises a clutch adapted to selectively connect a drive input to the transmission and frictional drive transmitting means comprising at least one drive disc having a frustoconical surface which engages frictionally with a drive annulus having a corresponding frustoconical drive surface, the disc and annulus being relatively movable between a position in which they are co-axial and a unity drive ratio is provided and a position in which the disc and annulus are eccentric and a different drive ratio is provided.

There are preferably two said discs mounted co-axially and trapping the drive annulus between their facing frustoconical surfaces. In this case it is necessary to provide some means for loading the discs towards each other; this may simply comprise a spring or may include a hydraulic loading device which varies the loading in accordance with the torque being carried by the transmission.

It is preferred to use a centrifugal clutch in the input to the variable speed transmission and further gearing may be provided to allow forward and reverse speeds and a neutral position to be selected. Thus this gearing may take the form of an epicyclic drive whose planet carrier may be connected to a fixed structure or to an output shaft.

The invention will now be particularly described merely by way of example with reference to the accompanying drawings in which:

Figure 1 is an axial cross-section through one embodiment of a transmission in accordance with the invention,

Figure 2 is a diagrammatic view of a control system for the transmission of Figure 1,

Figures 3 and 4 are diagrammatic views of alternatives to the control system of Figure 2,

Figure 5 is a view similar to Figure 1 but of an alternative embodiment,

Figure 6 is a section on the line 6-6 of Figure 5,

Figure 7 is a diagrammatic view of a control system suitable for the transmission of Figure 5 and,

Figure 8 illustrates diagrammatically how the drive transmission could be applied to a turbo-augmented piston engine.

In Figure 1 there is shown a drive transmission in which a bell casing 10 is connected to the end face of the cylinder block of a piston engine (not shown). The crankshaft 11 of the engine is drivingly connected to a fly-wheel 12 which has on its opposite face an annular driving surface 13. An input shaft 14 is located in a journal bearing 15 in a recess formed in the end of the crankshaft 11 and rotates co-axially with the crankshaft.

To allow drive to be selectively engaged between the input shaft 14 and the crankshaft 11 a clutch plate 16 having friction linings 16a is trapped between the drive surface 13 and a pressure plate 17. When the crankshaft 11 is stationary the diaphragm spring 18 acts between clutch housing 19 and the pressure plate 17 to withdraw the plate from the surface 13. There is thus no drive connection between the surface 13 and the plate 16.

Once the shaft 11 is driven by the engine to rotate at a speed above a pre-determined limit a series of centrifugal bob-weights 20 are fleuing outwards sufficiently to cause their levers 21 to overcome the force of the diaphragm spring 18 and to move the pressure plate 17 towards the surface 13 trapping the friction linings 17 of the disc 16. Thus drive is frictionally transmitted from the fly-wheel 12 to the disc 16. At its inner periphery the disc 16 is carried on a splined bush 22 which engages through splines 23 with the shaft 14. Therefore the overall effect of the arrangement is to drive the shaft 14 from the crank shaft 11 only when the crank shaft rotates at a speed in excess of a pre-determined minimum.

The shaft 14 protrudes through a seal 24 in an aperture 25 in the casing 10 into the interior of a main gear box casing 26 and within this casing it takes the form of a hollow shaft splayed out at its extremity to form a drive disc 27. The disc 27 is over-hung from the main bearing 28 which supports the shaft and is provided on its face remote from the bearing with a hardened drive surface 29 of frustoconical form.

Within the hollow centre of the shaft 14 there is located a secondary shaft portion 30. The shaft 30 is retained within the shaft 14 at one extremity by a pin 31 which extends through a hole in the shaft 30 and protrudes through axial slots 32 formed in the shaft 14. The pin 31 thus prevents relative rotational movement between shafts 14 and 30 but will permit a pre-determined degree of relative axial movement.

At its outer extremity adjacent the drive disc 27 the shaft 30 is formed to provide a second drive disc 33 which is in overall shape a mirror image of a shaft 27. There is thus provided a hardened frustoconical surface 34 which faces the surface 29.

The two shafts 14 and 30 and thus the discs 27 and 33 are urged into engagement by a stack of annular springs 35. These springs react against a fixed abutment 36 on the shaft 14 and a movable abutment 37 which may slide on the shaft 14 but which engages with the extremities of the pin 31. The spring 35 is arranged to be under compression so that it pushes the pin 31 to the left as shown in the

The drawings originally filed were informal and the print here reproduced is taken from a later filed formal copy.

drawings and consequently loads the surfaces 29 and 34 toward one another.

Engaging between the facing surfaces 29 and 34 of the disc 27 and 33 there is a drive annulus 38. This annulus has a thickened internal rim 39 having opposed frustoconical faces 40 and 41 whose shape conforms with, and which frictionally engage with the surfaces 29 and 34 respectively. In the position shown in full lines in Figure 1 the annulus 38 is concentric with the discs and therefore with the shaft 14.

At its outer periphery the annulus 38 has a series of splines 42 which engage with internal splines 43 on a flange 44 which extends axially from an output drive disc 45 which in turn forms the left-hand extremity of an output drive shaft 46. It will therefore be seen that drive from the shaft 14 will be frictionally transmitted from the discs 27 and 33 to the annulus 38 and thus through the splines 42 and 43 to the flange 44, the discs 45 and the output shaft 46. The shaft 46 is supported in two bearings 47 and 48 in a carrier 49 which is suspended through bushes 50 and 51 from a shaft 52. It will be appreciated that the entire carrier 49 together with the shaft 46 may be swung about the axis of the shaft 52 and in order to control its position a connection 53 is provided for the ram of a control mechanism as described below with reference to the later drawings.

In the position shown in Figure 1 drive between the shaft 14 and the output 46 will be transmitted at a unity ratio. However, if the carrier 49 is swung about the shaft 52 the axis of the shaft 49 will no longer coincide with that of the shaft 14 but there will be a pre-determined parallel displacement. This displacement will cause the annulus 38 to be displaced relative to the discs 27 and 33. Therefore although the drive input to the annulus 38 will continue to be at the same radius on the annulus the engagement between the discs and the annulus will take place at a smaller radius. Therefore a gear ratio of less than unity is provided, the exact value of the ratio depending upon the degree of displacement of the carrier 49 and thus the annulus 38.

Clearly in order that the annulus 38 should be able to move in this way while remaining in engagement with the discs 27 and 33 it is necessary that the discs should be able to move axially apart to accommodate the rim 39 between their facing surfaces and this is the purpose of the stack of springs 35. These springs allow this axial displacement while maintaining sufficient frictional force between the discs and the annulus to provide an effective transmission of the drive.

The apparatus so far described can clearly provide a stepless variation of gear ratio between unity and a lower figure dependent upon the dimensions of the various parts of the drive. In order to provide a facility for reverse and neutral positions, drive from the output shaft 46 is taken through an epicyclic gear train consisting of planets 54 which engage with an annulus gear 55 carried from the final output shaft 56 of the transmission. The planet carrier 57 is provided with an axially movable splined ring which engages with the outer surface of the carrier 57 and which may be caused to additionally engage with splines 59 formed on a fixed structure of the carrier 49 or alternatively with splines 60 formed on the outer periphery of the annulus gear 55.

It will be understood that if the splines on the member 58 engage with the splines 60 the epicyclic gear train is locked up and a direct drive will be provided between the output shaft 46 and the final output shaft 56. If the member 58 is in the position shown the shaft 46 is disconnected from the final output shaft 56 and this is effectively a neutral position. If the member 58 engages with the splines 59 the epicyclic drive operates to provide a reverse speed of slightly geared down ratio. To enable these three positions to be chosen a spring loaded ball 61 located in the carrier 57 may engage with one of three detents on the inner surface of the member 58 and an actuator 62 engages with the member 58 and may be slid along the shaft 52 by a selector 63 to move the member between its three detented positions.

It will be understood that as the carrier 49 is swung about the shaft 52 its rearward extremity which forms the output of the gear box will move relative to the static casing 26 and in order to prevent leakage of oil an elastomeric sealing member 64 extends between these two casings. A seal 65 is formed between the end of the carrier 49 and the shaft 56 to complete sealing of the transmission. Again the translating motion of the carrier 49 will produce a similar motion of the output drive flange 66 which forms the right hand extremity of the shaft 56. However this motion will be small enough to be accepted by the normal articulation of the propeller shaft of the vehicle.

The Figure 1 construction shows one further refinement in that a set of teeth 67 on the intermediate portion of the shaft 56 are arranged to co-operate with a pawl 68 which is controlled by means not shown to engage with the teeth only when the 'Park' position of the normally automatic gear selector is chosen. This then forms a transmission brake.

In Figure 2 there is diagrammatically illustrated a control system for the arrangement of Figure 1. The carrier 49 is shown supported from the shaft 52 and movable by a hydraulic cylinder or ram 70 which acts on the connection 53. The pressure of the hydraulic fluid in the cylinder 70 is provided by an oil pump 71 driven from a shaft 72 which is in turn driven from a gear 73 formed on the outer periphery of the bush 37. The shaft 72 is also arranged to drive a variable governor 74 having bob-weights 75 which act on a push rod 76 to operate a spring loaded spill valve. The output pressure from the oil pump 71 is carried in a duct 78 to the cylinder 70 and a spill duct 79 carries the fluid to the spill valve 77. In addition to being operated by the governor 74 the spill valve is acted on via a spring 80 by a lever 81 which in turn is decided on by cams 82 and 83. The cam 82 is linked to the position of the accelerator pedal of the vehicle while the cam 83 is linked to the normal transmission gear selector which is operable by the driver.

In operation therefore the pressure in the spill duct 79 and hence in the duct 78 is controlled by the valve 77 in accordance with actual speed of the engine, the gear selected by the driver and the position of the accelerator pedal. These latter two parameters act

on the lever 81 in a highest wins manner so that only the position of the over-riding one of the two will contribute to control the pressure.

It will generally be seen that if the governor 74 exceeds a speed determined by the loading placed on the spring 80 the valve 77 will allow the pressure in the line 79 to drop and will therefore allow the piston of the ram 70 to retract and to increase the gear ratio of the transmission. An increase in the selected speed on the gear box will have the same effect as will an increased demand transmitted by the accelerator pedal. The control system therefore provides a control which takes these parameters into account. There is one further feature of the Figure 2 system in that the oil spilt from the valve 77 is allowed to flow through a duct 84 to feed lubrication ducts 85 within the gear box.

The Figure 2 device is a hydromechanical arrangement and it may prove simpler in the future to use an electronic control which may include micro-processor circuitry. Figure 3 shows a layout of control system which would utilise such a miniaturised computation device. Once again a ram 70 is provided with hydraulic pressure from an oil pump 71 driven from a shaft 72 and gear 73. In this case, however, the duct 85 which transmits fluid from the pump to the ram has an over-pressure relief valve 86 and a one-way valve 87. There are also two spill ducts 88 and 89 which are controlled by solenoid operated valves 90 and 91 respectively. The valves are normally open valves which are energised to close by signals from a computer 92 which receives inputs related to the engine speed, the gear selector position and the accelerator pedal position.

Of course the computer 92 would be pre-programmed to provide the necessary control characteristics but the provision of the two valves 90 and 91 and the one-way valve 87 enables three separate conditions of operation to apply. If neither valve is energised both will be open and the pressure in the line 85 will be low. The ram 70 will retract causing the gear ratio to increase toward direct drive, or alternatively the governed speed of the engine will reduce.

In the second condition where both are energised the pressure in the line 85 will increase. This will either increase the governed speed or hold the transmission in its lowest ratio. The third condition arises where only valve 90 is energised. The valves 90 and 87 prevent any escape of oil from the ram 70 and in this condition it is therefore possible to hold an intermediate ratio.

In Figure 4 there is shown a further and simpler electronic control system. Once again the pump 71 provides oil to the ram 70 and a computer 92 is used to give the necessary control characteristic. In this case, however, the line 95 between the pump and the ram has only a single spill duct 96 controlled by a single spill valve 97 which again may be energised by the computer 92 to close. Here again three conditions of operation are available. If the valve 97 is not energised to close pressure in the ram 70 will increase to reduce the governed speed or to hold direct drive. If the valve is energised, it closes and increases the pressure in the line 95 to increase the governed speed or to hold the lowest gear ratio. In order to provide hold at an intermediate ratio it is necessary either to hold the valve 97 or to enegise it sufficiently to spill at the intermediate pressure.

Referring next to Figure 5 this illustrates a second embodiment of the mechanical layout of the variable ratio transmission. In this case a centrifugal clutch basically similar to that described above is used and the description is not repeated. Drive from the centrifugal clutch is taken to an input shaft 100 which again passes through a seal 101 in the bell casing. In this case the shaft 100 is formed with a driving disc 102 at an intermediate position along its length, the disc having a hardened frustoconical face 103 similar to the face 29 of Figure 1. The shaft 100 is mounted in a ball bearing 104 adjacent to the disc 102 and it extends beyond the disc 102 to slide within a second hollow shaft 104. The shaft 104 is splined to the shaft 102 by axial splines 105 and the shafts 100 and 104 are both supported by bearing 104 and a further roller bearing 107. The shaft 105 is provided at its left-hand extremity with a disc 108 which is similar to the disc 102 but faces in the opposite direction. Thus the disc 108 has a frustoconical face 109 which is hardened and which is opposed to the face 103. In order to urge the disc 102 and 108 together a stack of springs 110 are provided which engage with an end piece 111 to push the shaft 105 to the left and with the housing 112 which reacts the loads back to the shaft 100.

In the present case as well as the springs 110 hydraulic pressure is used to urge the discs together and to this end the end plate 109 is sealed by a piston ring seal 113 to an axially extending annular flange 114 which extends forwardly from the piece 112. Tappings 115 in the disc 105 supply hydraulic fluid to the space thus formed from a supply arrangement to be described below. Inbetween the discs 112 and 108 there engages a drive annulus 116. This annulus is shown in Figure 5 as being concentric with the discs and it has frustoconical end faces 117 and 118 which engage with the faces 103 and 109 of the discs. These faces are similarly hardened. As in the previous case the annulus 116 must be moved so that its axis moves parallel to the axis of the shaft 100 so as to vary the ratio of the frictional drive between the discs and the annulus. Therefore the annulus is supported in roller bearings 119 and 120 in a yoke member 121 which is in turn supported in journal bearings 122 and 123. Actuation rod 124 has a gear rack 125 which engages with a corresponding arcuate rack 126 on the yoke 121 so that the yoke and the annulus may be moved in an arc about the axis of the bearings 122 and 123. This provides the necessary translational movement of the annulus whilst maintaining its axis parallel with the shaft 100.

To take the drive from the annulus 116 it is provided on its outer periphery with a ring of gear teeth 126 which engage with an output gear 127 formed on an output shaft 128. The shaft 128 is mounted in a ball bearing 129 and surrounds a final output shaft 130 supported in bearing 131. Thus the bearings 129 and 131 support between them both the shafts 128 and 130.

Drive from the gear 127 is taken through the shaft

128 to a further gear 132 which engages with a series of planet gears 133. The planet gears in turn engage with an internal ring gear 134 formed on a flanged disc 135 which is carried from the shaft 130. Once again the planet carrier 136 is provided with a sliding engagement member 137 which is splined to its outer surface and which will in turn engage with the internal annulus gear 134 or part of the fixed structure at 138. This again provides a forward reverse or neutral drive.

As mentioned above the loading on the disc 102 and 108 is provided by hydraulic means as well as by the springs 110. Supply of hydraulic fluid is via a floating coupling 140 to a central bore 141 in the shaft 100 and then through radial bores 142 to an annular sealed space 143 between the shafts 100 and 105 and hence through the drillings 115 to the space within which the springs 110 are located.

Figure 6 shows how the actuating rod 124 is operated under the two influences of a pinion 145 which is driven by an actuator not visible in this view and a hydraulic piston 146 provided with loading pressure through a drilling 147. Operation will be disclosed with reference to Figure 7 below.

Additional features to note in the mechanical arrangement of Figure 5 are that a pawl 150 may be provided to engage a rack 151 on the other surface of the disc 135 to provide a transmission lock similarly to that provided by 68 in Figure 1. A further ratchet 152 may engage with a rack 153 on a fixed structure of the gear box. This ratchet is arranged to disengage above a certain speed but it will operate as a hill holder device. Finally a third ratchet with pawls 154 engaging with a rack 155 is provided in the centrifugal clutch so as to enable tow starting of the engine at low speeds. Again this ratchet is disengaged by centrifugal force above a predetermined engine speed.

Figure 6 shows diagrammatically the control system used to provide the hydraulic load pressure for the discs 102 and 108 and which operates the actuator rod 125. Once again a computer 160 is provided with inputs related to engine speed, vehicle speed, engine torque and gear selector position and it has outputs to an actuator 161 and a solenoid valve 162. The actuator 161 drives the pinion 145 through a worm and wheel arrangement 163 and hence operates the rod 124 to vary the position of the yoke 121. In this way the ratio of the drive is steplessly varied.

The valve 162 operates to control the pressure of hydraulic fluid provided by a pump 164 to the balance piston 146 and to the chamber formed inside the sleeve 114. The computer is pre-programmed to vary the valve 162 in such a way as to provide a pressure within the sealed space inside the sleeve 114 which would be sufficient in conjunction with the spring load of the springs 110 to provide a contact force in the frictional drive arrangement just sufficient to enable drive to take place but not sufficient to provide excessive losses. The same pressure is applied to the piston 146 and by making the total displacement of these two systems equal the total flow requirement from the pump 164 is only that required to replenish leakage. The pump 164 may therefore be small and consume very small power and in addition the forces in the system are nearly balanced out so that a relatively small force is required on the actuation rod 124 and the motor 161 need not be very large.

The embodiment of Figures 5 and 6 is clearly very compact and it may be used for front wheel or rear wheel drive since it has the advantage that there is not lateral movement of the output drive such as is present in the Figure 1 embodiment.

It will be noted that since the gear transmission is stepless it is very suitable for use in conjunction with a turbo-charged engine in which the high inertia turgo-charger rotor is normally a problem.

Figure 8 shows diagrammatically how a device such as that shown in Figures 5 and 6 may be used in conjunction with a turbo-charger. In this case the engine shown at 170 drives a centrifugal clutch 171 and a variable drive transmission 172 which will be similar to that of Figure 5. Exhaust from the engine passes from the manifold 173 through a turbine 174 and the turbine shaft 175 drives through a stepless variable ratio drive 176 which will be seen to be similar to that of Figure 1 but will of course be scaled down and simplified. Output from the variable ratio drive is taken by gear wheels 177, 178 and 179 to enhance the output of the engine.

This arrangement may be used to give very high expansion ratios while overcoming the problems of variation in turbine speed otherwise necessary and rotor inertia in the case of governor transmissions which are other than stepless. It will also be noted that by using the friction drive as the first speed reduction from the turbine the noise and wear problems of very high speed toothed gear are avoided.

CLAIMS

1. An automotive transmission comprising a drive input, a clutch, a frictional drive transmission and an epicyclic drive transmission, the clutch being adapted selectively to connect the drive input to the frictional drive transmission which comprises at least one drive disc having a frustoconical surface and a drive annulus having a corresponding frustoconical drive surface which engages frictionally with the surface of the drive disc, the disc and annulus being relatively movable between a position in which they are coaxial and a unity drive ratio is provided between them and positions in which the disc and annulus are eccentric and a different drive ratio is provided, the epicyclic drive transmission being connected to receive the output of the frictional drive transmission and comprising a driving sun wheel, a plurality of planet wheels engaging with the sun wheel, a plant carrier carrying the planet wheels, an output annulus engaging with the planets, and engagement means having three modes, in a first of which it engages between said carrier and said output annulus so that said epicyclic drive is locked-up, in a second of which it frees said carrier so that no drive is transmitted by the drive, and in a third of which it engages between said carrier and fixed structure so that reverse drive of lower ratio is transmitted by the epicyclic drive.

2. An automotive transmission as claimed in claim 1 and in which there are two said drive discs mounted coaxially and trapping said drive annulus

between their facing frusto-conical surfaces.

3. An automotive transmission as claimed in claim 2 and in which said drive discs are urged together to provide a load on the engaging disc and annulus faces sufficient to produce the frictional force necessary to allow transmission of drive.

4. An automotive transmission as claimed in claim 3 and in which said discs are urged together by springs.

5. An automotive transmission as claimed in claim 3 and in which said discs are urged together by a hydraulic load.

6. An automotive transmission as claimed in claim 5 and comprising a hydraulic piston-and-cylinder arrangement connected so that its internal volume depends upon the relative positions of the axes of the discs and annulus, this internal volume forming a compensatory part of the hydraulic system urging the discs together and thus enabling the system to operate with only small overall changes in volume.

7. An automotive transmission as claimed in claim 2 and in which there is an input shaft to the frictional drive transmission upon which said two drive discs are mounted.

8. An automotive transmission as claimed in claim 7 and in which there is an output shaft from the frictional drive transmission extending parallel with but not overlapping the input shaft and carrying at one end said drive annulus which is overhung to lie between the drive discs, the output shaft being carried in a carrier which is movable so as to displace the axis of the output shaft to a desired position in relation to that of the input shaft.

9. An automotive transmission as claimed in claim 8 and in which there is an articulated drive shaft which carries drive from the output of the transmission, the articulated of the drive shaft being sufficient to allow the output shaft perform its required displacement.

10. An automotive transmission as claimed in claim 7 and in which there is rotatable yoke which carries said drive annulus so that the annulus can rotate about its own axis and be swung about the axis of rotation of the yoke, and a lay shaft mounted coaxial with the yoke and carrying a gear which engages with a set of gear teeth formed on an outer surface of said ring so as to transmit drive from said ring to said layshaft.

11. An automotive transmission as claimed in claim 10 and in which said layshaft is hollow and an output shaft for the transmission is located coaxially within the layshaft, the output shaft carrying said output annulus of said epicyclic transmission at one end and said layshaft carrying said driving sun wheel of said epicyclic transmission at its corresponding end.

12. An automotive transmission as claimed in claim 1 and comprising a selectively operable pawl and a set of teeth formed on an output shaft of the transmission, the pawl when operated engaging with said teeth to provide transmission braking.

13. An automotive transmission as claimed in claim 1 and in which there is a centrifugally disabled pawl carried from a shaft between such clutch and said frictional drive and a set of teeth on fixed structure of the transmission, the pawl when operative engaging with said teeth to prevent rotation of the input shaft and hence to provide a hill-holder device for the driven vehicle.

14. An automotive transmission as claimed in claim 1 and in which said clutch has a centrifugally disabled pawl carried from its normally driving parts and a set of teeth on its normally driven parts, the pawl engaging with the teeth when operative to allow tow-starting at low speeds of the engine of the driven vehicle.

15. A control system for an automotive transmission of the kind incorporating a frictional variable drive transmission having at least one drive disc with a frustoconical surface and a drive annulus with a corresponding frustoconical drive surface which engages frictionally with the surface of the drive disc, the disc and annulus being relatively movable by actuation means between a position in which they are coaxial and a unity drive ratio is provided between them and positions in which they are eccentric and a different drive ratio is provided, the control system comprising a hydraulic system which operates the actuation means which comprises a hydraulic mechanism.

16. A control system as claimed in claim 15 and comprising a centrifugal bob-weight governor which operates a valve to control the pressure of hydraulic fluid delivered to said actuation means in accordance with the rotational speed of the input to the transmission.

17. A control system as claimed in claim 16 and comprising a spring loading on said governor which is biassed in accordance with a speed demand signal from a driver-operated accelerator control and a signal from a driver-operated gear selector.

18. A control system as claimed in claim 17 and in which said signals comprise the degree of rotation of cams operating on a lever which acts on said bias.

19. A control system as claimed in claim 15 and comprising an electronic computer device and at least one valve controlled by said computer to affect the pressure of said hydraulic fluid in accordance with control parameters of the driven vehicle and of the engine.

20. A control system as claimed in claim 19 and comprising a pump, a supply duct from said pump to said actuation means, a non-return valve in said supply duct, first and second bleed ducts connected to said supply duct upstream and downstream respectively of said non-return valve, and first and second valves controlled by said computer and controlling the flow from said first and second bleed ducts respectively.

21. A control system as claimed in claim 15 and in which hydraulic pressurising means are provided to urge said disc and annulus together to provide a friction-producing load between the contacting surfaces, the hydraulic pressure from said control system to said actuation means also being fed to said pressurising means.

22. A control system as claimed in claim 21 and in which the areas of the surfaces on which said hydraulic pressure acts in said pressurising means and

said actuation means are arranged so that the effect of the hydraulic actuation means and that of the hydraulic pressurisation means substantially cancelled out, and further electric actuation means are provided, controlled by an electronic computer so as to provide the small force necessary to move said disc and annulus to their desired relationship.

23. An automotive transmission comprising an internal-combustion engine having an exhaust-driven turbocharger, the turbocharger comprising an exhaust-driven turbine, a variable speed frictional transmission and a drive connection from the transmission to the engine output shaft, the variable speed transmission comprising at least one drive disc having a frusto-conical surface and a drive annulus having a corresponding frustoconical drive surface which engages frictionally with the surface of the drive disc, the disc and annulus being relatively movable between a position in which they are coaxial and a unity drive ratio is provided between them and a position in which they are eccentric and a different drive ratio is provided.

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